HARDY (J.K.)

DSIR, LONDON

THE CONDENSATION OF WATER
ON REFRIGERATED SURFACES

DEPARTMENT OF SCIENTIFIC AND INDUSTRIAL RESEARCH

FOOD INVESTIGATION

SPECIAL REPORT No. 54

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BY

J. K. HARDY

K. C. HALES

AND

G. MANN

(Ditton Laboratory)

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July, 1951
Department of Scientific and Industrial Research,
Charles House, 5–11 Regent Street,
London, S.W.1.

FOREWORD

THE experimental work described in this report was started early in 1938 when Mr. Hardy and Mr. Mann investigated heat and water transfer to or from, and the skin friction of, a flat wetted plate cooler. In May 1938 the need for experimental work on the design and performance of commercial pipe bank coolers was discussed at the Consultative Group (Refrigerating Plant) of the Food Investigation Organization and experimental work on this problem was started by Hardy, Hales and Mann. This work was continued until 1942 and, in addition to the work on a standard commercial type cooler, observations were made on a pipe bank cooler constructed of 1 in. diameter light-weight tubing and on two coolers (constructed of 1 in. and 1½ in. diameter tubing) of the type used to cool the 'tween deck spaces of cargo vessels.

The two latter units were provided by Messrs. J. & E. Hall Ltd., who also seconded two members of their staff to assist with the experimental work in which these coolers were used.

Work on the present report was not begun until 1946. The delays in its preparation have been unavoidable owing to the separation of the three authors and their pre-occupation with other tasks. Mr. Hardy had joined the Royal Aircraft Establishment in 1940. Mr. Hales had transferred to the staff of Refrigerated Cargo Research Council in 1946. The Department is indebted to these two authors for the work they have done, since they ceased to be members of the staff of D.S.I.R., in the preparation of the report. To Mr. Mann has fallen the task of preparing the various drafts.

FRANKLIN KIDD
Director of Food Investigation

FOOD INVESTIGATION ORGANIZATION,
20A Regent Street,
Cambridge.

AUTHORS' ACKNOWLEDGEMENTS

ASSISTANCE in carrying out the experimental work which forms the basis of this report was given by Messrs. R. Morris and G. W. Goodwin of the Ditton Laboratory and by Messrs. R. T. Hales and J. W. Taylor of Messrs. J. & E. Hall Ltd., of Dartford.

Messrs. J. & E. Hall provided the commercial-type coolers used for the experiments described in Part III of the report.

CONTENTS

THE CONDENSATION OF WATER ON REFRIGERATED SURFACES

GENERAL INTRODUCTION

THE rate at which water vapour from a stream of air will condense on a cold surface is of importance both in the transport and storage of food and in the control of the humidity of air. In the case of food, the rate at which heat is transferred to the refrigerated pipes is increased because of the transfer of latent heat by evaporation from the food and condensation subsequently on the surface of the pipes. A problem in the storage of the more perishable foodstuffs is to reduce the loss of water to a minimum.

In the transport of food, a problem of particular interest is concerned with the effect on the performance of the cooler of frost, which accumulates on the surface of the pipes. The frost insulates the pipes thermally, and at the same time changes the geometrical arrangement of the cooler and increases its resistance to the flow of air. This causes the performance of the cooler to deteriorate progressively, the rate of deterioration in a particular design being dependent upon the rate at which frost grows and on its insulating properties.

The problem, generally, of predicting the rate at which vapour from a stream of air will condense on a surface was analysed in 1937[(1)]* and is briefly summarized in Part I of this Report. It is shown that a very simple relationship exists between the coefficient of heat transfer from air to a dry surface and the coefficient of mass transfer when there is condensation from the air stream on the surface. The validity of this method of analysis has been tested by experiments on a single wetted plate and on banks of pipes. These experiments form the subject of Parts II and III of the present report. They cover a range in conditions of operation. In the case of a pipe bank, measurements of the rate of heat transfer and of resistance to air flow were made with the surface of the pipes dry. This provided the basic data required to analyse the performance when the pipes were wet, and allowed also a comparison to be made with the data of Grimison[(2)]. Measurements of rate of condensation of water vapour and of transfer of heat were made in conditions such that the surfaces of the pipes were wet with water. Finally, a general study was made of the frosting of the bank.

The method of analysis, though directed more particularly towards the specific problem of the condensation of water in the refrigeration of air, is of wider application. It is of interest that it has been applied successfully by Hardy[(3)] to the problem of protecting aircraft against ice by heating the exposed surfaces. In this connection, it has been applied also to predict the rates of evaporation and the temperature in a system with two components, alcohol and water. It can be used, quite generally, in problems involving the transport of matter by diffusion in a stream of gas, and has the merit that measurements of the transfer of heat can provide the basic data.

*** These numbers refer to references on page 29.**

Symbol	*Meaning*	*Units*
A_p	area of cross-section	sq. ft.
C_p	specific heat of air	B.Th.U./lb.°F.
D	diffusivity of vapour	sq. ft./sec.
d	diameter	ft.
d_h	hydraulic diameter $\left(=\dfrac{4\times \text{Area}}{\text{perimeter}}\right)$	ft.
e	vapour pressure	in. mercury
G	weight flow per unit area	lb./hr. sq. ft.
g	gravitational constant	ft./sec.^{2}
H	rate of transfer of heat	B.Th.U./hr. sq. ft.
h	conductance of heat	B.Th.U./hr. sq. ft. °F.
K	thermal conductivity of frost	B.Th.U.ft./hr. sq. ft. °F.
k	,, ,, ,, air	B.Th.U.ft./hr. sq. ft. °F.
k_f	coefficient of surface friction	no dimension
k_h	,, ,, transfer of heat	,, ,,
k_p	,, ,, resistance	,, ,,
k_w	,, ,, transfer of water	,, ,,
L	latent heat of vapourization	B.Th.U./lb.
N	number of rows of pipes	
n	concentration of vapour	lb./lb. of dry air
Nu	Nusselt's Number $\dfrac{Hd}{k\theta}$	no dimension
P	barometric pressure	in. mercury
Pr	Prandtl's Number $\dfrac{Cp\mu}{k}$	no dimension
Δp	pressure drop	lb./sq. ft.
R	Reynold's Number $\dfrac{Vd\rho}{\mu}$	no dimension
Tr	Taylor's Number $\dfrac{\mu}{D\rho}$	no dimension
t	temperature	°F.
V	velocity	ft./sec.
W	rate of condensation	lb./hr. sq. ft.
w	rate of flow by weight	lb./hr.
y	latent heat as fraction of total	
μ	viscosity of air	lb./sec. ft.
ρ	density of air	lb./cu. ft.

Subscripts

m	mean
o	main stream
s	surface
t	total

PART I.—THEORETICAL CONSIDERATIONS

by

J. K. Hardy

The process to be analysed is the transfer, by diffusion of vapour, of water from a stream of air to a surface which is cooled to below the dew-point of the air. Because the surface is colder than the air there will be, at the same time, a transfer of heat by convection from the air to the surface. The mechanism of transfer both of heat and matter is related, through Reynolds analogy in its extended form, to that of the transfer of momentum which causes surface friction.

The analogy between heat transfer and surface friction has been confirmed by experiments in which the flow was through pipes having smooth walls. It fails if the walls are rough, and also if the surfaces are of such a shape as to cause intense gradients of pressure. The analogy may be extended to cover the case of the transfer of matter by diffusion and, as the nature of the process is identical with that of the transfer of heat, neither the shape nor the roughness of the surface, so far as is known, impose any restriction. It is possible therefore to predict the rate of transfer of matter from the rate of transfer of heat or the reverse.

RATE OF CONDENSATION

The rate at which water from air flowing at velocity V will condense on unit area of a surface may be expressed by the equation

$$W = k_w \rho V (n_o - n_s) \times 3600^* \qquad \text{(1)}$$

In this equation, k_w is a non-dimensional coefficient which will be called the coefficient of transfer, since it applies irrespective of whether the process is one involving evaporation or condensation. The potential in the process of transfer is $n_o - n_s$, namely, the difference between the concentration of vapour in equilibrium with the water, or ice, at the surface and that in the air outside the boundary layer.

In the case of a battery of pipes, or of flow through a pipe, the value of n_o is the average concentration in the stream as a whole. The concentration is expressed as the mass of vapour per unit mass of air. It may be expressed in terms of vapour pressures by substitution from an equation which is derived from the gas equation, viz.

$$n = 0{\cdot}622 \frac{e}{P-e} \qquad \text{(2)}$$

in which e is the vapour pressure and P the barometric pressure. Equation (1) then becomes

$$W = k_w \rho V \left(\frac{e_o}{P - e_o} - \frac{e_s}{P - e_s} \right) \times 0{\cdot}622 \times 3600 \qquad \text{(3)}$$

In which e_s is the vapour pressure at saturation at the temperature of the surface. The vapour pressure, usually, is so small relative to the barometric

* The inclusion of 3600 in this and subsequent equations is to convert to hour units.

pressure that P may be substituted for $(P-e)$ and equation (3) may be written

$$W=k_w \rho V \frac{(e_o-e_s)}{P} \times 0{\cdot}622 \times 3600 \qquad (4)$$

The value of k_w, is fundamentally determined by the thickness of the boundary layer and by the nature of the flow, as defining the field of diffusion, and by a parameter which gives a measure of the activity of the process of diffusion. This parameter is called Taylor's Number (Tr) and is defined as

$$Tr=\frac{\mu}{D\rho} \qquad (5)$$

in which D is the diffusivity of the vapour, and μ and ρ are the viscosity and density of the air, or other gas, through which the vapour is diffusing.

RELATION BETWEEN COEFFICIENTS OF EVAPORATION AND OF TRANSFER OF HEAT

The coefficient of transfer of water, k_w, is related through Reynold's analogy to k_h, the coefficient of transfer of heat. The equation which defines k_h is

$$H=k_h \rho V C_p (t_o-t_s) \times 3600 \qquad (6)$$

or in terms of h the unit conductance,

$$h=k_h \rho V C_p \times 3600 \qquad (7)$$

so that, in relation to Nusselt's Number, $\frac{hd}{k}$

$$k_h=\frac{Nu}{Pr.R} \qquad (8)$$

Reynold's analogy is between heat transfer and surface friction. Both are processes of diffusion, of heat in the one case, and of momentum in the other. The ratio of the diffusivities, known as Prandtl's Number (Pr), is

$$Pr=\frac{C_p \mu}{k} \qquad (9)$$

The equations which relate k_h with k_f, the coefficient of surface friction, were derived by Goldstein[4]. These equations, with substitution of k_w for k_h and of Tr for Pr, apply to processes of condensation or evaporation.

When the flow is laminar

$$k_h=k_f(Pr)^{-\frac{2}{3}} \qquad (10)$$

$$\text{and } k_w=k_f(Tr)^{-\frac{2}{3}} \qquad (11)$$

$$\text{So that } \frac{k_w}{k_h}=\left(\frac{Pr}{Tr}\right)^{\frac{2}{3}}=\left(\frac{D\rho C_p}{k}\right)^{\frac{2}{3}} \qquad (12)$$

When the flow is turbulent, the equation corresponding to (10) is

$$\frac{1}{k_h}=\frac{1}{k_f}+5{\cdot}6(Pr-1)\sqrt{\frac{1}{k_f}} \qquad (13)$$

So for turbulent flow,

$$\frac{k_w}{k_h}=\frac{1+5{\cdot}6(Pr-1)\sqrt{k_f}}{1+5{\cdot}6(Tr-1)\sqrt{k_f}} \qquad (14)$$

In evaluating this equation for air, the value of k_j can be taken, with sufficient accuracy, as a little less than that of k_h.

TOTAL RATE OF TRANSFER OF HEAT

When the temperature of a cold surface is lower than the dew-point temperature, the total rate at which heat is transferred is the sum of the rates of transfer by convection and by evaporation. The total rate of transfer H_t is

$$H_t = H + LW \qquad \ldots \qquad (15)$$

In this, L is the latent heat, which will be the sum of the latent heats of vaporization and fusion if the pipes are at a temperature such that ice forms.

Substitution for H from equation (6) and for W from equation (4) gives

$$H_t = k_h \rho V C_p (t_o - t_s) + L\, k_w \rho V \frac{(e_o - e_s)}{P} \times 0{\cdot}622 \qquad \ldots \qquad (16)$$

The diffusivity of water vapour in air, it appears, is almost exactly the same as that of heat, as will be discussed later. It follows that the value of k_w, practically, is the same as that of k_h. Writing k_h for k_w in equation (16), and substituting from equation (7) gives

$$H_t = h\ [(t_o - t_s) + \frac{L}{C_p}\ \frac{(e_o - e_s)}{P} \times 0{\cdot}622] \qquad \ldots \qquad (17)$$

This equation, and equation (16) also, applies only when the surface temperature of the pipes is below the dew-point temperature of the air. When there is no condensation, the total rate of transfer is given by equation (6).

RATIO OF k_w TO k_h FROM THE PSYCHROMETRIC EQUATION

The wet-bulb thermometer registers a temperature such that the heat received from the air by convection, apart from a small loss by radiation, balances exactly the heat lost by evaporation from the bulb. There is convection because the temperature of the wet bulb is below that of the air, which is registered by the dry-bulb thermometer. There is evaporation because the vapour pressure at the surface of the bulb is greater than that in the air. The psychrometric equation is equation (16) with $H_t = 0$, because, on the balance, there is no gain or loss of heat by the wet bulb. So from equation (16)

$$e_o = e_s - \frac{P\, C_p}{0{\cdot}622 L}\ \frac{k_h}{k_w}\ (t_o - t_s) \qquad \ldots \qquad (18)$$

The temperature t_o and t_s are those of the dry and of the wet bulb respectively.

The psychrometric equation which has been derived empirically from the results of experiments, and from which the psychrometric tables are derived, is given by Brooks[5] as

$$e_o = e_s - 0{\cdot}000652 P (t_o - t_s)\ (1 + 0{\cdot}00102\ t_s) \qquad \ldots \qquad (19)$$

for pressures in millimetres and temperatures in degrees centigrade. In order to make equation (18) identical with equation (19) the value of k_h/k_w,

at 32°F., must be 1·004. It appears, therefore, that Taylor's Number for water vapour in air is almost identical with Prandtl's Number. The value of Prandtl's Number for air at 32°F. is 0·714. To make Taylor's Number equal to this, the diffusivity of water vapour must be $1{\cdot}96 \times 10^{-4}$ sq. ft./sec. (0·182 sq. cm./sec.), a value which is somewhat lower than that given in the International Critical Tables. The value deduced from the psychrometric equation is probably the more reliable, so that the value of k_h/k_w should be taken as 1·0 at 32°F. The value changes slowly with temperature; at 0°F. it is 0·996 and at 60°F. is 1·007.

EFFECT OF SIZE OF COOLER ON RATE OF CONDENSATION

To some extent, the rate at which water vapour will condense in the cooler as a whole can be increased or diminished by changes in the surface temperature. The temperature at which the surfaces are designed to operate can be changed by increasing or diminishing the total area of surface in the cooler. If the area is sufficiently large for the surfaces all to be at a temperature above that of the dew point of the incoming air, which makes e_s greater than e_o in equation (17), there will be no condensation. If the incoming air is unsaturated, t_o can be greater than t_s so that there will be a transfer of heat without condensation. What can be achieved in the direction of preventing condensation depends very much on the humidity of the incoming air, and on the extent to which it is practicable to increase the size of the cooler. The humidity of the air is conditioned by the character of the material being stored, and the rate of heat leakage into the store. So in an actual case, it is necessary to calculate the rate of evaporation from the material as well as the rate of condensation on the cooler, as the processes are mutually dependent.

The fraction of the total heat which represents the latent heat of condensation, from equation (17) is

$$y = \frac{\frac{L}{C_p P}(e_o - e_s)}{(t_o - t_s) + \frac{L}{C_p P}(e_o - e_s)} \qquad \ldots \quad (20)$$

The optimum surface temperature in a particular cooler may be determined from this equation. The equation has been evaluated in Figure 1 for the conditions shown.

In designing for a low rate of condensation, the efficiency of the cooler as a heat exchanger is important. An increase in the conductance of heat from air to surface has the same effect as an increase in area in increasing the temperature of the surface. The relation between geometrical arrangement and performance is discussed fully by Hardy.[6]

HEAT TRANSFER FROM PIPE TO BRINE

When brine or other liquids are used as cooling agents the heat transferred to the exposed surface of the pipes is transferred in turn to the brine. The rate of transfer may be calculated from the equation

$$Nu = 0{\cdot}023\ Pr^{0{\cdot}4} \left(\frac{wd_h}{A_p \mu}\right)^{0{\cdot}8} \qquad \ldots \quad (21)$$

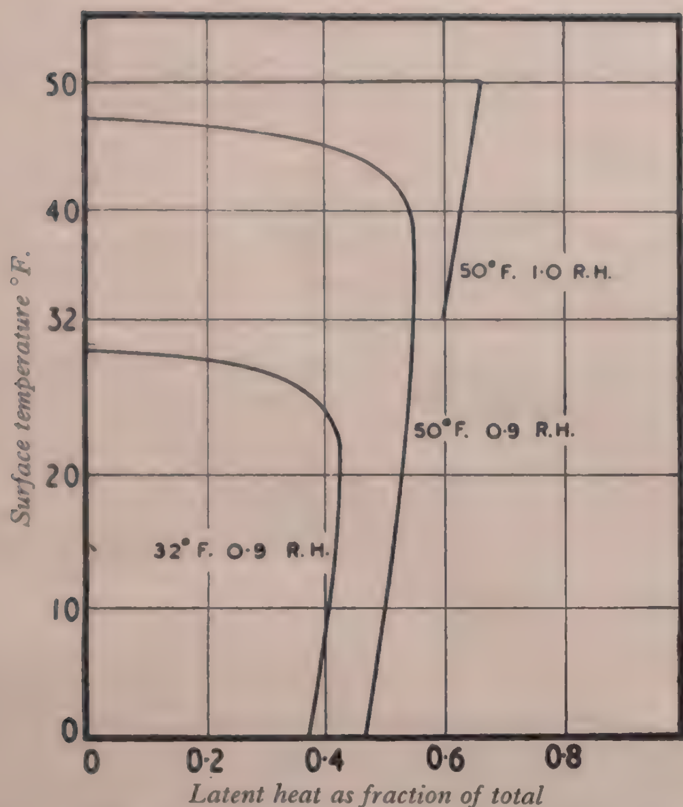

FIG. 1. *Effect of surface temperature on rate of condensation*

or from equation

$$k_h = 0{\cdot}023\ Pr^{-0\cdot6} \left(\frac{wd_h}{A_p\mu}\right)^{-0\cdot2} \quad \ldots \quad (22)$$

In practice, equation (22), in conjunction with equation (7), seems to be the easier to use.

These equations apply to pipes or passages of any section, provided hydraulic diameter is used, as shown in the equation.

The overall conductance from air to brine, denoting these by subscripts $_1$ and $_2$, is given by the equation

$$h_{1,2} = \frac{1}{\dfrac{1}{h_1} + \dfrac{d_1}{d_2\,h_2}} \quad \ldots \quad (23)$$

This gives the conductance per square foot of the outer surface of the pipe. The thermal resistance of the walls is omitted as being negligible compared with the resistance between brine and pipe, and pipe and air.

EFFECT OF FROST

Frost, which accumulates by sublimation on to the surfaces of the cooler, insulates these surfaces thermally, and also affects the performance of the cooler by changing its geometry.

In the case of a pipe battery, the geometrical arrangement of the pipes in terms of pipe diameter is changed. So, also, is the variable of primary

importance in determining performance, namely the weight flow of air per unit area of the spaces between the pipes. The effect of these changes on the performance of the cooler may be calculated from the data of Grimison.[2]

The insulating effect of frost depends on its porosity, and therefore on its density. The density varies from the surface inwards owing to diffusion and re-sublimation within the frost. At present, it is only possible to treat the frost as though it had a uniform insulating value irrespective of depth. As will be shown in the section which deals with the experiments, the insulating value depends on the rate of deposition and on other factors.

The equation which gives the overall conductance from air to brine through a frost covered pipe, per square foot of exposed surface of the frost, is

$$h_{1,3} = \frac{1}{\frac{1}{h_1} + \left(\frac{d_1 - d_2}{2K}\right)\frac{d_1}{d_m} + \frac{1}{h_3}\frac{d_1}{d_3}} \qquad \text{(24)}$$

in which K is the thermal conductivity of the frost, and d_m is the mean diameter. Strictly

$$d_m = \frac{d_1 - d_2}{\log_e \frac{(d_1)}{(d_2)}}$$

but the thickness of the frost is such that the arithmetical mean usually is sufficiently accurate. The subscripts, $_1$, $_2$ and $_3$, denote the outer surface of the frost, the outside of the pipe and the inside respectively. Equation (24) is the equation derived from the basic equation for insulated pipes as given by McAdams [7], p. 12.

RESISTANCE OF COOLER

The resistance of a cooler to the flow of air depends both on the density of the air and on its rate of flow. It is convenient to use a coefficient of resistance, k_p, which is a function only of Reynold's Number, and of the geometrical arrangement of the battery.

In the case of a battery of pipes, the coefficient of resistance is defined by the equation

$$k_p = \frac{\Delta p \; g}{\rho V^2 N} \qquad \text{(25)}$$

In this, Δp is the loss of total head in lb./sq. ft. through the battery, N the number of rows of pipes, and V is the velocity at the point of maximum constriction in each row. If the total area at maximum constriction is A_p, and w is the rate of flow by weight

$$\rho V = \frac{W}{A_p} = G \qquad \text{(26)}$$

So, by substitution

$$k_p = \frac{\Delta p g \rho}{N G^2} \qquad \text{(27)}$$

When the change in temperature of the air in passing through the cooler is large, the measured resistance will differ appreciably from the actual resistance because the air loses momentum. The reverse occurs in the part of the circuit in which the air receives heat. The equations necessary to calculate the actual resistance from that measured are given by McAdams[7].

PART II.—THE TRANSFER OF WATER VAPOUR BETWEEN AN AIR STREAM AND A WETTED PLATE IN RELATION TO THE TRANSFER OF HEAT

By

J. K. Hardy and G. Mann

INTRODUCTION

It is difficult, by direct measurement, to determine with accuracy the rate at which water condenses on a surface. The rate of condensation, and the rate at which heat is transferred, can be measured indirectly by measuring the distribution of vapour pressure, temperature, and velocity in the boundary layer of retarded air near the surface. These measurements are taken at various stations along the surface, and the rates of transfer are found by integration. The intensity of fluid friction at the surface may be found by the same process. This indirect method of measurement has the advantage that the rates of transfer locally at different positions along the surface may be determined.

This method has been used in a series of experiments on a single wetted plate mounted vertically in a small wind tunnel. A film of fluid maintained on the plate was cooled by a horizontal tangential air stream.

Earlier experiments had been made by Hardy[1] with a wet-cooler which consisted of a bank of plates with brine flowing down them. The results were unsatisfactory, principally because the distribution of flow over the plates was uneven. The data from the single plate may be used in the design of a multi-plate cooler if allowance is made for the establishment of pipe-flow in the passages between the plates.

APPARATUS

The general arrangement is shown in Figure 2. A glass plate 3 ft. high and 4 ft. long was used. The thickness was $\frac{1}{8}$ in., and at the leading edge this was reduced to $\frac{1}{64}$ in. by a long bevel on the opposite side to that on which the measurements were taken. The leading edge was rounded so that the air should flow smoothly along the sides of the plate.

Both water and brine were used in the experiments and particular care was taken to obtain a uniform flow over the entire plate. The rate of flow was 1 lb./min. per ft. run down each side of the plate. The temperature of the fluid supplied to the plate was controlled to within 0·2°F. by a thermostat.

The tunnel was installed in an insulated chamber in which temperature and, to some extent, humidity could be controlled.

MEASUREMENTS

Measurements of velocity, temperature and vapour pressure were taken at five stations along the plate in the air as it flowed past. The stations, which were in line with the centre of the tunnel, were distant 0·84, 1·57, 1·96,

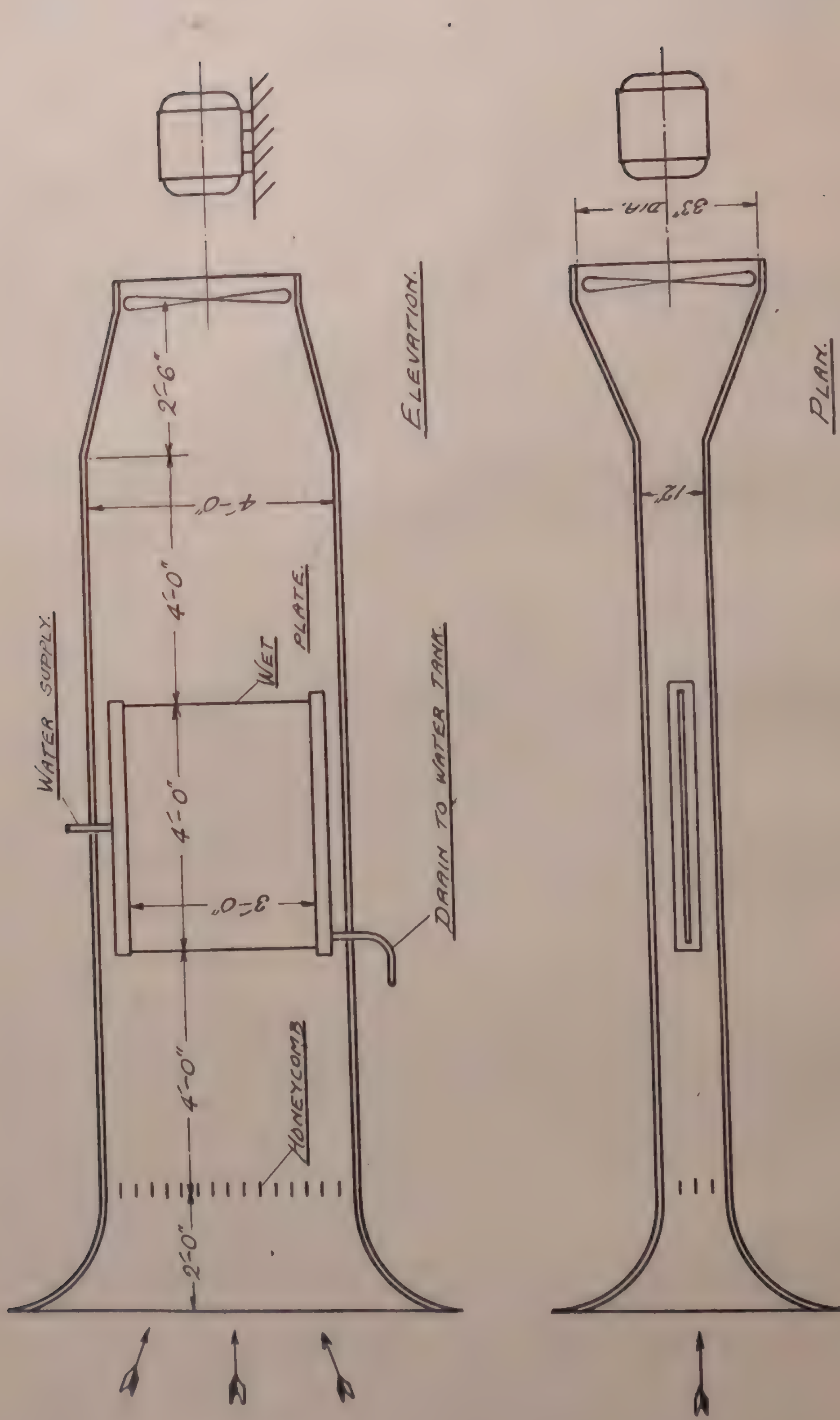

FIG. 2. *Wind tunnel with wet plate*

2·97, and 3·81 ft. from the leading edge of the plate and will be designated stations 1 to 5. At each station sufficient measurements were taken to establish the profile of velocity etc. in the boundary layer.

Velocity was measured with a pitot tube made from 0·4 mm. hypodermic tubing in conjunction with a static tube of standard design set midway between the plate and the walls of the tunnel. An arrangement for traversing the tube was provided and this was mounted on a stiff frame so that there could be no movement relative to the plate. The position of the tube was registered by a dial gauge.

Temperature in the boundary layer was measured differentially with respect to that in the main stream by two thermocouples of 40 S.W.G. wire. The method of mounting the wires is shown in Figure 3. The wire which traversed the boundary layer was aligned accurately, so as to be parallel with the plate and square with the direction of flow of air in order to avoid a

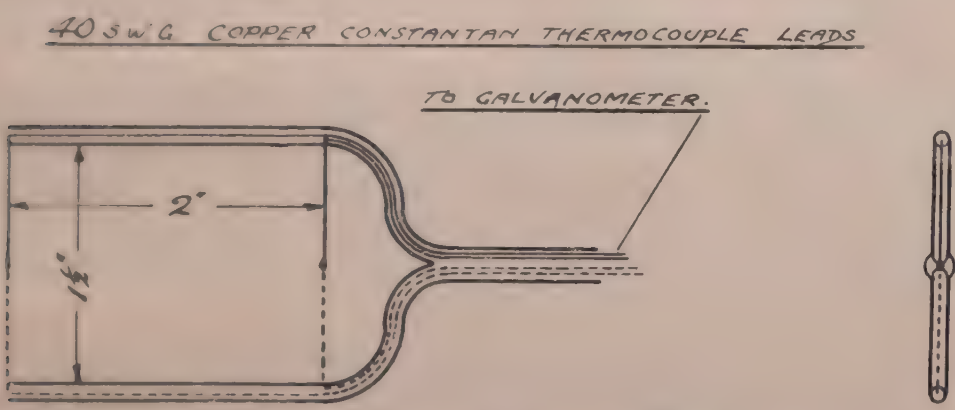

FIG. 3. *Thermocouple arrangement for measuring air temperatures in boundary layers relative to free air stream*

gradient in temperature along the wire. It was possible, by just submerging the wire in the water on the surface of the plate, to measure direct the difference between the temperature of the water next to the surface and that of the air in the main stream. The temperature of the water was measured separately by a submerged thermocouple.

Vapour pressure was measured by a dew-point apparatus which was specially constructed for the purpose. A sample of air was withdrawn from the boundary layer through the same hypodermic tube as was used for the measurement of velocity. The rate of removal was adjusted so that the air entered the tube always at a velocity less than that locally in the boundary layer. In this way, it is believed, a true sample, locally, was obtained without disturbing the normal distribution of vapour in the boundary layer.

The dew-point instrument consisted of a small polished disc of Monel metal which was cooled from below by a jet of paraffin directed squarely at the centre of the disc. The space above the disc, through which the sample of air was drawn, had a volume of about 0·06 cu. in. This is sufficiently small for one sample to be swept out completely and rapidly by another of

different composition, and sufficiently large for the readings to be unaffected by the deposition of a trace of water on the mirror. The temperature of the mirror was measured by a thermocouple which was centrally placed. Because the dew formed initially, and disappeared finally from the surface immediately around the position of the thermocouple, it was possible to obtain remarkably consistent readings. The average difference in temperature between the appearance and the disappearance of the dew was 0·2°F.

The measurements of velocity, vapour pressure and temperature in the boundary layer were taken in sequence so that the same equipment for traversing could be used for each.

ACCURACY OF MEASUREMENTS

The difference between pitot and static pressure was measured to ±0·0003 in. W.G. at the lower air speeds. At higher speeds, the fluctuations in pressure made the accuracy somewhat inferior.

The difference in temperature between main stream and boundary layer was measured with an accuracy of ±0·01°F.

The accuracy of the dew-point instrument was checked by measuring the dew-point of air which was saturated at a known temperature. Air from a bottle partly filled with water was displaced through the instrument by means of water from another bottle. Both bottles were completely immersed in a bath which was held at a constant temperature and was stirred continuously. The temperatures of the air in the bottle and of the mirror of the instrument are shown in Table 1.

TABLE 1. *Test of Dew-point Instrument*

Temp. of air in bottle °F.	Temp. of mirror °F.	Difference °F.
42·92	42·80	0·12
51·18	51·22	0·04
58·55	58·55	0·00
60·40	60·34	0·06
60·65	60·60	0·05
62·80	62·70	0·10

From the average difference in temperature between air and mirror, it appears that the dew-point registered by the instrument is 0·055°F. below the true dew-point. This, at the temperatures in the experiments on the plate, corresponds to an error in vapour pressure of 0·00073 in.

Distances on the traverser were measured to within ±0·0025 in. The distance from the surface of the liquid on the plate is not known to this degree of accuracy because of fluctuation in the thickness of the stream. The typical variation in thickness as a result of many observations is 0·002 in., the average thickness being 0·012 in. at the level at which the measurements were made. Distances, in the case of temperature and vapour pressure, are to the geometric centre of the thermocouple and of the sampling tube. In measuring velocity, the distance is to the effective centre of the tube as calculated by Stanton.(11)

The maximum velocity of the air in the tunnel was limited by the blowing off of water from the plate. This occurred at a speed of 34 ft./sec. The velocities used in the experiments were 19, 23·5 and 32·5 ft./sec.

The experiments for the most part were concerned with the condensation of water. In some experiments the temperatures were changed so that water evaporated from the surface of the plate.

The range covered is shown in tabular form below.

TABLE 2. *Range of experiments*

Velocity ft./sec.	19		23·5		32·5	
Temp. air (°F.)	57.2		55.4		59·0	
Temp. water (°F.)	37·4		37·4		38·5	
Vp air (in. Hg)	0·35*E**	0·35*D**	0·366*D*	0·492*D*	0·453*D*	0·315*E*
Vp water (in. Hg)	0·472*E*	0·236*D*	0·264*D*	0·315*D*	0·236*D*	0·393*E*

* *D*. Condensation on water surface.
E. Evaporation from water surface.

RESULTS

The gradients of velocity, temperature and vapour pressure in the boundary layer may be compared directly by plotting, against distance from the plate, the value of $\frac{V}{V_o}$, $\frac{t-t_s}{t_o-t_s}$, and $\frac{e-e_s}{e_o-e_s}$. The local values at any position in the boundary layer are V, t and e. The subscripts $_o$ and $_s$ indicate the value in the main stream and at the surface of the liquid on the plate respectively.

A typical set of profiles of distribution is shown in Fig. 4 for five stations along the plate. The rates of transfer were deduced by integrating the profiles of distribution and so finding the change between one station and the next. The intensity of surface friction was found from the change in total head. The rate of transfer of heat and of condensation, respectively, were found by plotting $V\frac{(t-t_s)}{t_o-t_s}$ and $V\frac{(n-n_s)}{n_o-n_s}$ against distance from the surface, and integrating graphically. The value of n, the concentration of water vapour in lb./lb. of air, is given by equation (2) p. 3.

The coefficients of transfer are given on page 15.

The values for the coefficients are the average values between stations, and in evaluating Reynolds Number there is difficulty in assigning an appropriate value to the length. If, for instance

$$k_h \propto L^n$$

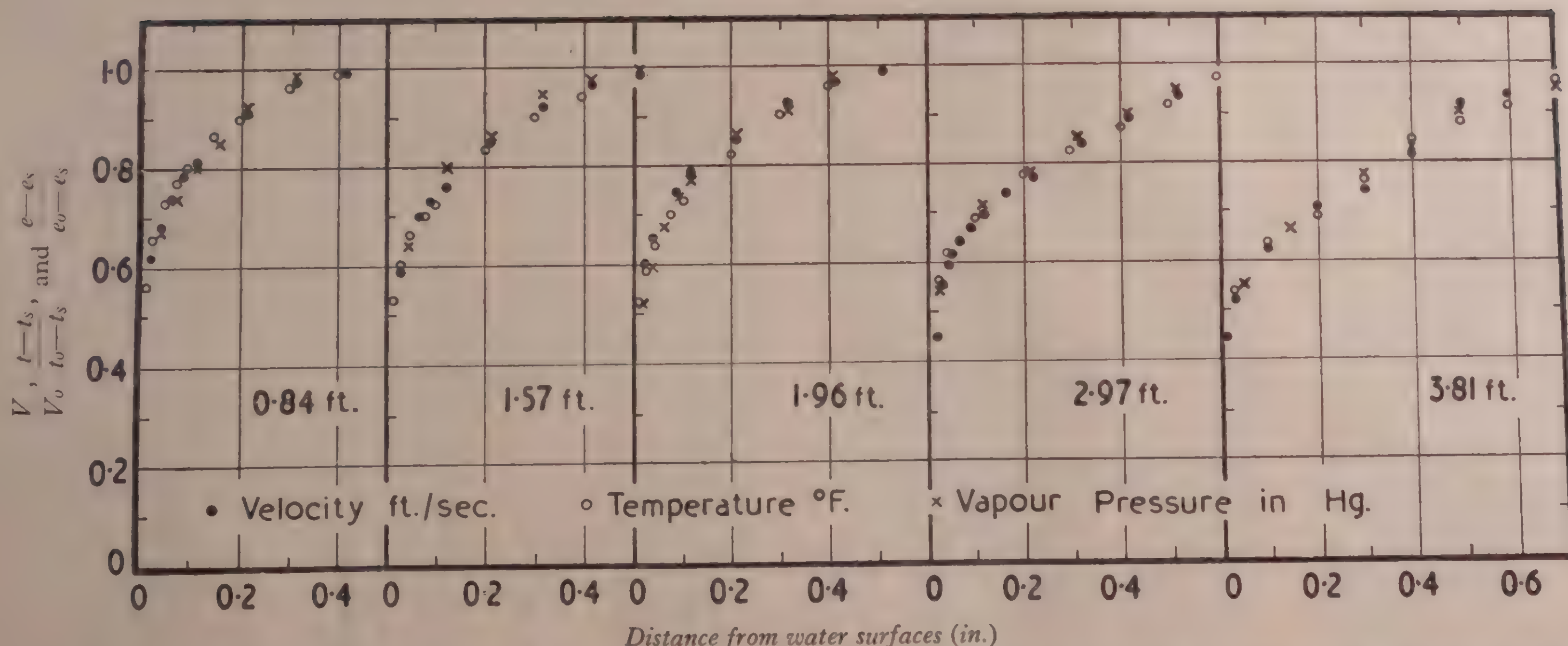

FIG. 4. *Profiles of velocity, temperature, and vapour pressure in boundary layer at various distances from the leading edge*

TABLE 3. *Coefficients of transfer*

Station	Velocity ft./sec.	Log $_{10}$ of Reynold's No.	k_f dry	k_f wet	k_h	k_w	k_h/k_w
0–1	19·0	4·69	0·0046	0·0049	0·0046	0·0051	0·90
0–1	23·5	4·78	—	0·0045	0·0044	0·0044	1·0
0–1	32·5	4·93	0·0045	0·0043	0·0044	0·0043	1·02
1–2	19·0	5·15	0·0032	0·0028	0·0026	0·0025	1·04
1–2	23·5	5·24	0·0031	0·0025	0·0026	0·0025	1·04
1–2	32·5	5·38	0·0031	0·0024	0·0027	0·0028	0·963
3–4	19·0	5·46	0·0025	0·0027	0·0028	0·0026	1·07
3–4	23·5	5·55	—	0·0029	0·0025	0·0025	1·0
4–5	19·0	5·70	0·0022	0·0027	0·0021	0·0021	1·0
4–5	32·5	5·83	0·0021	0·0024	0·0021	—	—

it can be shown that the mean length, for stations 1 and 2, is

$$L_{1.2} = \sqrt[n]{\frac{L_2^{n+1} - L_1^{n+1}}{(n+1)(L_2 - L_1)}} \quad \ldots \quad (28)$$

The values of n have been found by trial, and Reynold's Number has been evaluated by using the mean length as defined above.

DISCUSSION

The average of the values of k_h/k_w given in Table 3 shows that k_w equals k_h exactly. There is a scatter in the individual values owing both to errors of observations and to errors in the process of integration. The equality in the rates of diffusion of heat and of water vapour is shown more directly by the similarity in the profiles of distribution.

The values of both k_h and k_w may be in error to a small extent owing to the fact that the temperature at the actual surface of the liquid on the plate cannot be measured. The point of measurement was below the surface, so that the difference in temperature, and hence in vapour pressure, between surface and air as measured is somewhat greater than it should be. The values of k_h and k_w may therefore be too small. The relative values of k_h and k_w are practically unaffected as can be shown by recalculating the values after making an arbitrary adjustment in the values of t_s.

An error of approximately 5 per cent of the value of $(t_a - t_s)$ is suggested by the difference between the average observed value of k_f/k_h and the average theoretical value from equation (13) p. 4. The difference may to some extent be caused by the film of water being blown back from the leading edge of the plate. The thermal boundary layer therefore starts after the friction layer has formed.

The primary interest is the relation between the rates of transfer of heat and of water vapour and this is unaffected by the displacement of the film on the leading edge. It is of interest that consistent and satisfactory results can be obtained by indirect measurements.

There was no change in the values of k_h and k_w when temperatures were reversed and water was evaporated from the plate. This would not be so had

conditions been such as to cause water to condense as droplets in the air of the boundary layer. Theoretically, the total rate of transfer of heat h_t of equation (17) p. 5, should not be affected by the formation of fog in the boundary layer, because, as shown by the experiments, water vapour and heat diffuse at the same rate. So, the smaller gradient in vapour pressure caused by condensation should be balanced exactly by the larger gradient in temperature. Random movement of the droplets of water within the boundary layer would cause some slight departure from exact equality.

PART III.—HEAT TRANSFER FROM AN AIR STREAM TO BANKS OF PIPES

by

K. C. Hales

INTRODUCTION

The main purpose of the experiments with banks of pipes was to compare, under different conditions, the actual rate at which water condensed on the surface of the pipes with that predicted by calculation. The temperature of the pipe surface in these experiments was above the freezing point. In addition, a series of experiments was made with the pipe surface at lower temperatures, so that condensation was in the form of frost. These latter experiments, which were exploratory, were made in order to find out the way in which the performance of the cooler was affected and how the density of the frost was affected by changes in the conditions under which the cooler was operating.

Several different designs of cooler were used. For the most part these were experimental in character and consisted of separate grids which were mounted so that the connecting bends and headers were completely screened from the air stream. The use of separate grids enabled changes to be made to the geometry of the cooler; screening of bends and headers avoided short circuiting of the air. Experiments were made also with two coolers of the type used on board ship to cool spaces of about 10,000 cu. ft. capacity. These had different sizes of pipes and were tested with the object of comparing their frosting characteristics. Of equal interest, however, is the comparison of the performance of these coolers with that predicted by Grimison[2]. His data were derived from experiments on small banks of tubes arranged in a regular geometrical pattern and spanning the duct from wall to wall. Commercial coolers differ from this idealized arrangement because the connecting bends between pipes and the headers interfere with the regular geometry and allow air to short circuit round the cooler. There is also considerable short circuiting through the space between the cooler and the condensate tray. The effect of this is to reduce the performance of the cooler as a heat exchanger to below that which would be predicted from results obtained with an indealized design. The quantity of air which short circuits the cooler has been determined indirectly so it is possible to suggest tentatively the allowance which should be made when Grimison's data are used.

Measurements of the transfer of heat when the pipes were dry have been made for all the experimental coolers. As stated earlier, the experimental coolers were of the idealized type with straight runs of pipe only exposed to the air stream. The pipes were of galvanized steel as used in commercial coolers and were cooled by brine. It is of interest therefore to compare the coefficients of transfer of heat with those measured by Pierson[8] who used small diameter pipes with smooth walls, heated electrically. The coefficient of transfer of heat from the brine to the walls of the pipe has been measured

for some of the coolers. The resistance of the coolers to flow of air, and in one case to the flow of brine, also has been measured.

APPARATUS

The coolers used in the experiments have been mentioned briefly in the previous section; the particulars of their design are summarized in Table 4.

The conductance of sensible heat was measured for all these coolers, condensation of water was measured on the first three only, and the effect of frost was measured on all but the first two. The first five of the coolers

TABLE 4. *Particulars of Coolers Tested*

Cooler	Arrangement	Spacing of pipes (diameters) Transverse Lateral	No. of Rows	No. of pipes per row	Pipe Diameter (in.)	Length pipe (ft.)	Surface Area (sq. ft.)
I	Square	2·1 × 2·1	3	8	1·91	66	33
II	Square	2·1 × 2·1	6	8	1·91	132	67
III	Square	2·1 × 2·1	17	8	1·91	380	190
IV	Square	2 × 2	10	12	1·125	330	97
V	Triangular	2 × 2	10	12	1·125	330	97
VI	Triangular	2·25 × 1·44	38	9	1·91	1610	810
VII	Triangular	2·76 × 1·68	52	10	1·34	1930	680

were of the idealized form, already mentioned, and were mounted in a closed circuit wind tunnel 2 ft. 9 in. wide and 2 ft. 8 in. high. The grids were mounted vertically with the axis of the pipes horizontal. The grids were coupled in series so that the direction of the flow of the brine from grid to grid was the same as that of the air. Brine was circulated through the cooler in a closed circuit, the temperature being held constant by injection of a colder brine. The heat removed by the cooler from the air in the tunnel was restored by an electrically heated grid of wires. During experiments in which condensation was taking place, the humidity of the air was maintained by the injection of steam. The point of injection was chosen so that mixing would be as complete as possible. The general layout of the apparatus is shown in Figure 5. The tunnel was situated in a refrigerated chamber so that insulation of the tunnel was unnecessary, the temperature of the air in the chamber being adjusted so as to equal that of the mean temperature of the air in the tunnel.

The size of Coolers VI and VII precluded the use of a return circuit so they were tested as fan and cooler units, the chamber itself being the return circuit. Modifications had to be made to the brine circuit and to the means used to return heat and water vapour to the air.

METHOD OF MEASUREMENT

Temperature

Temperatures were measured by copper-constantan thermocouples. The accuracy of the measurement of e.m.f., for which a potentiometer

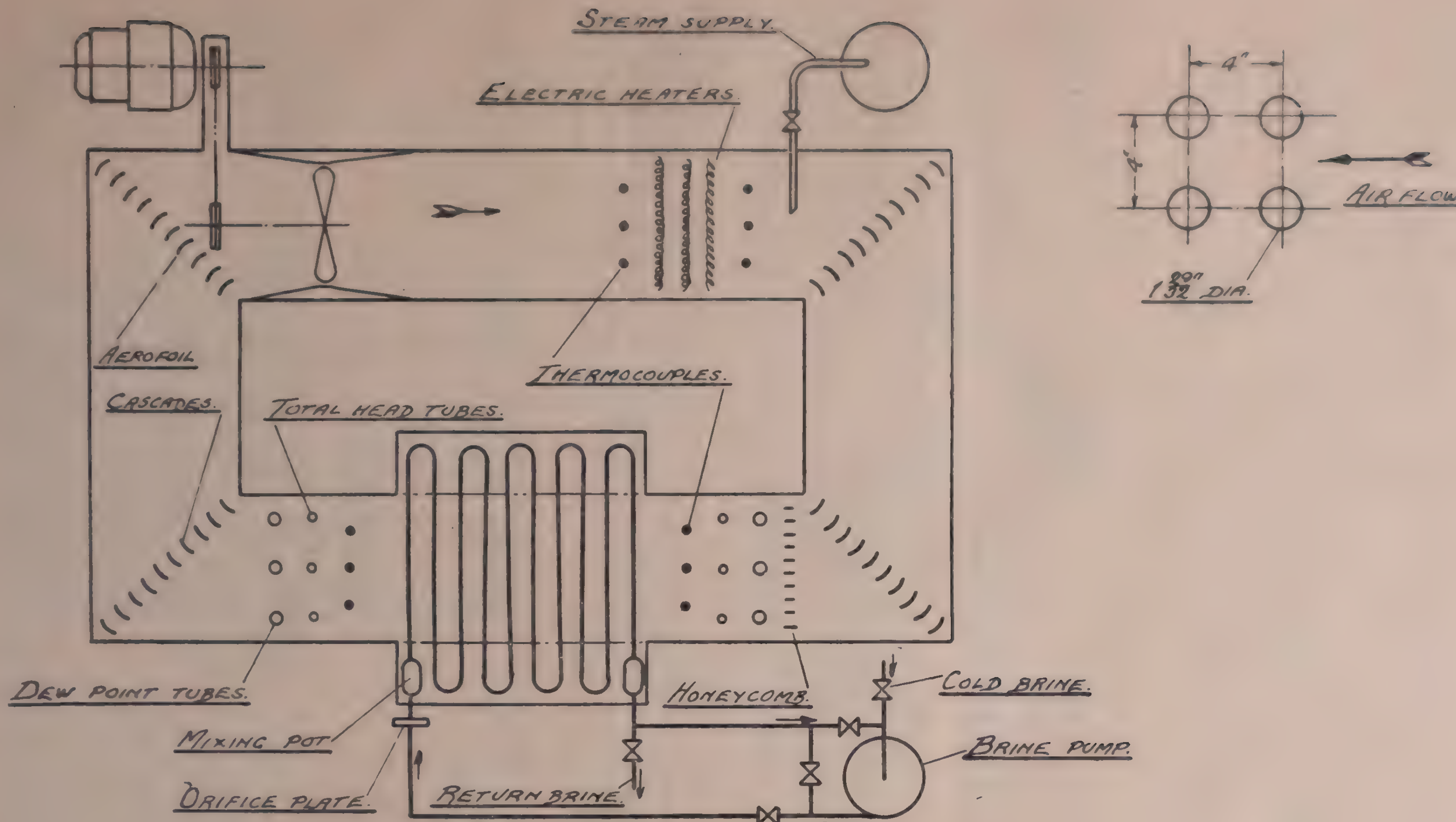

FIG. 5. *Arrangement of tunnel for testing pipe batteries*

was used, was $\pm 1\mu V$ which corresponds to a temperature of $\pm 0{\cdot}05^\circ$F. Precautions were taken to ensure that the measured temperature was in fact the true temperature. The methods adopted were as follows:

(i) *Brine Temperature.* In order to obtain the true average temperature the brine was passed through a mixing device, illustrated in Figure 6, immediately upstream of the thermocouple. No measurable difference of temperature could be detected in the stream of brine after mixing.

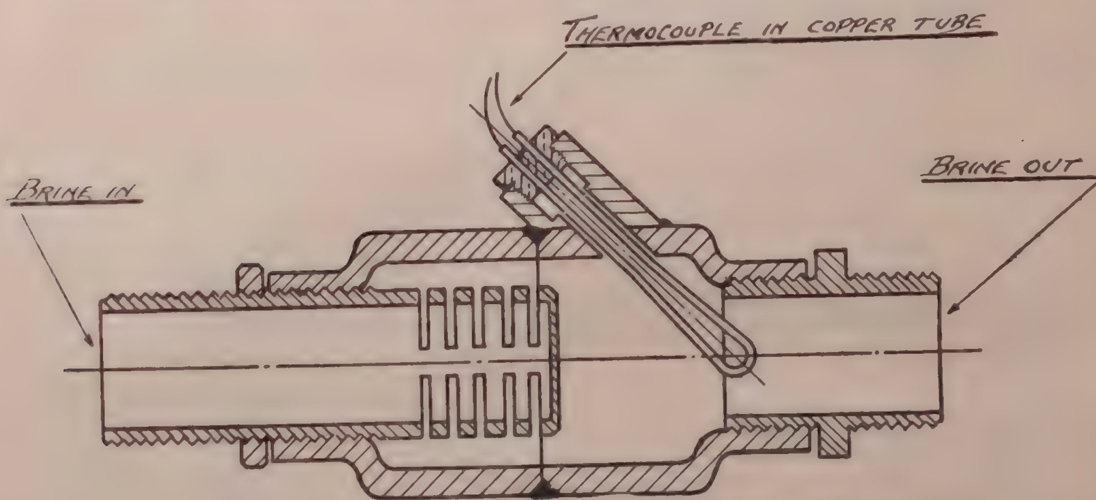

FIG. 6. *Mixing pot for measuring brine temperatures*

(ii) *Pipe Surface Temperature.* The temperature of the pipe wall was measured by thermojunctions of 40 S.W.G. wire. The junctions were soldered into shallow grooves which were filled flush with the surface. The position of these thermocouples in Cooler II is shown in Figure 7. The thinness of the pipe in Coolers IV and V and the design of coolers

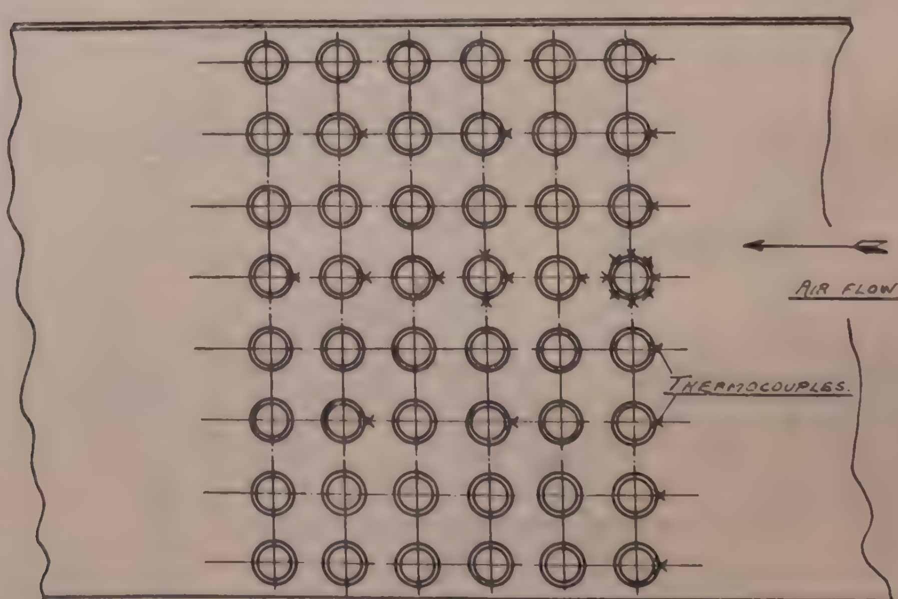

FIG. 7. *Positions of thermocouples on cooler II*

VI and VII precluded the direct measurement of surface temperature by this method. For these coolers the surface temperature was calculated by using equation (22) p. 7 to calculate the conductance from brine, and using the measured values of brine temperature.

(iii) *Air Temperature*. The temperature of the air was measured by nine thermocouples placed so that each one was the centre of an area one ninth that of the cross section of the tunnel. They were situated 1 ft. in front of the first row of pipes in the cooler and 2 ft. behind the last row. An exploration of the air stream in the vicinity of these stations showed that the maximum deviation in temperature was less than 5 per cent of the mean.

Air Flow and Resistance

The rate of flow of air was measured by nine total head tubes, in the same position as the thermocouples, in conjunction with static-pressure holes in the walls of the tunnel in the same plane. It was determined also by heating a grid of wires electrically at a known rate and measuring the increase in the temperature of the air across the grid.

In the case of Coolers VI and VII, velocity was measured by traversing the inlet duct with a pitot-static tube or, alternatively, with a vane anemometer.

The resistance of the coolers to the flow of air was determined by measuring the difference in total head, the nine tubes at the inlet being used in conjunction with similar tubes at the outlet.

Brine Flow

The rate of brine flow was measured by an orifice plate. The orifice plate was calibrated *in situ* and the calibration checked from time to time by diverting the brine into a weighing tank.

Humidity

The humidity of the air was measured by an aspirated wet- and dry-bulb psychrometer, using thermocouples for measuring temperature. The accuracy was $\pm 0{\cdot}05^\circ$F. which corresponds to an accuracy in vapour pressure of $\pm 0{\cdot}00045$ in. in the range of temperature occurring in the experiments, namely 32°–40°F.

METHOD OF EXPERIMENT

Transfer of Sensible Heat

In the series of experiments to determine the coefficients of transfer of sensible heat, the temperature of the brine was adjusted so that there was no condensation in any part of the cooler. In these experiments the temperature of the pipe surface was in the range of 35°F. to 55°F., and that of the air in the tunnel from 40°F. to 60°F. The weight flow of air per unit area at the point of maximum constriction between pipes in the experiments was from 1,200 lb./sq. ft. hr. to 10,000 lb./sq. ft. hr.

Measurements were made of the conductance of heat from the pipes to the brine, during the course of the experiments on Cooler III, for various rates of flow from 150,000 lb./sq. ft. hr. to 300,000 lb./sq. ft. hr.

Measurements were made also of the resistance of the coolers to the flow of air, and of the grids to the flow of brine.

Transfer of Latent Heat

In this series of experiments, the brine temperature was adjusted so that water was condensing on the entire surface of the cooler. The range in rates of air flow was the same as for the preceding experiments. The rate of condensation varied from 0·025 to 0·1 lb./sq. ft. hr.

The rate of condensation was determined indirectly by measuring the total rate of transfer of heat and deducting the component of sensible heat which was calculated from the data of the first series of experiments. The total rate was determined by measuring rate of flow and temperature change of the brine. Measurements of the surface temperature of the pipes gave the information required to calculate the transfer of sensible heat. This was found to be preferable to determining sensible heat by direct measurement from the temperature change and flow of air through the cooler, because the temperature change through the cooler was small and the air temperature at outlet irregular.

RESULTS

In deriving the coefficients of transfer, the logarithmic mean of the difference in temperature, or vapour pressure, between the air and the pipe surface was used. The physical properties of the air were evaluated at the arithmetical mean of the temperatures of the air and the pipe surface.

Values of the coefficient of transfer of sensible heat, k_h and of resistance, k_p, are given in Table 5. For comparison, the values of the coefficient derived from Grimison's data[2] are also quoted.

A selection of the results from experiments to determine the coefficient of transfer of latent heat are given in Table 6. The experiments chosen are representative, and the quantities given are sufficient to show the average conditions which obtained during the course of each experiment.

TABLE 5. *Coefficient of Transfer of Sensible Heat from Air to Pipes*

Cooler	Weight flow/ unit area lb./ sq. ft. hr.	Mean Air Temp. (°F.)	Mean Surface Temp. (°F.)	Rate of Heat Transfer B.Th.U./ sq. ft. hr.	k_h	k_h from Grimison[2]	k_p	k_p from Grimison[2]
I	1950	51·0	34·8	75	0·0099	0·0096	0·12	0·12
I	8550	63·0	39·6	264	0·0057	0·0052	0·12	0·12
II	7500	59·6	47·1	147	0·0067	0·0065	0·12	0·11
II	1500	63·1	45·0	78	0·0121	0·0120	0·11	0·11
III	1950	52·8	40·8	62	0·0112	0·0116	0·11	0·11
III	7460	62·0	54·2	73	0·0070	0·0069	0·10	0·11
IV	1820	53·0	37·4	90	0·0141	0·0145	0·08	0·11
IV	5550	48·3	39·8	114	0·0113	0·0097	0·10	0·11
V	1710	50·0	38·3	65	0·0156	0·0170	0·18	0·20
V	7000	50·0	40·0	119	0·0094	0·0098	0·15	0·16
VI	2700	39·0	35·5	22·6	0·0100	0·011	0·14	0·18
VI	6100	43·0	38·0	53	0·0071	0·0082	0·13	0·16
VII	2600	29·5	25·3	27	0·0112	0·0125	0·15	0·18
VII	6200	43·0	37·6	60	0·0078	0·0087	0·13	0·15

TABLE 6. *Coefficient of Transfer of Latent Heat from Air to Pipes*

Cooler	Weight Flow lb./sq.ft. hr.	Mean Air Temp. (°F.)	Mean Air Vapour Pressure in. Hg.	Mean Surface Temp. (°F.)	Sensible Heat Transfer B.Th.U./sq. ft. hr.	Latent Heat Transfer B.Th.U./sq. ft. hr.	k_h	k_w	$\frac{k_h}{k_w}$
I	1760	60·6	0·403	36·6	106·0	79·5	0·0102	0·0107	0·95
I	1780	60·7	0·410	36·7	107·0	80·5	0·0102	0·0104	0·98
I	2880	59·8	0·39	37·3	134·0	100·0	0·0084	0·0081	0·96
I	3670	54·2	0·362	37·0	121·0	102·0	0·0086	0·0086	1·00
I	4590	55·6	0·398	40·3	124·0	108·0	0·0070	0·0074	0·94
I	4590	55·8	0·402	40·3	124·0	123·0	0·0070	0·0080	0·88
II	1375	51·0	0·332	37·7	54·0	41·0	0·0119	0·0121	0·98
II	1380	51·0	0·315	37·5	53·5	35·0	0·0119	0·0108	1·10
II	1700	49·0	0·315	37·3	54·0	40·0	0·0116	0·0112	1·04
II	2060	46·2	0·279	34·9	59·5	43·0	0·0110	0·0121	0·91
II	3300	44·1	0·272	35·9	58·0	44·0	0·0090	0·0098	0·92
II	9100	48·9	0·307	42·0	88·0	48·0	0·0060	0·0061	0·99
II	9150	48·6	0·303	41·6	88·5	47·0	0·0060	0·0060	1·00
II	9500	49·9	0·315	42·7	93·0	46·8	0·0059	0·0055	1·07
III	2400	44·4	0·248	35·5	64·0	27·0	0·0107	0·116	0·92
III	2970	47·0	0·283	38·1	61·0	40·0	0·0099	0·0106	0·93
III	2980	47·4	0·283	37·4	67·0	41·0	0·0099	0·0100	0·99
III	3120	47·5	0·276	38·3	62·0	39·0	0·0096	0·0097	0·99
III	3200	47·2	0·279	37·6	67·0	32·0	0·0096	0·0089	1·07
III	3700	44·1	0·244	35·5	70·0	30·2	0·0090	0·0100	0·90
III	4850	42·6	0·256	36·2	71·5	39·4	0·0081	0·0082	0·99
III	5500	43·3	0·256	36·1	77·0	34·2	0·0078	0·0063	1·23
III	5800	43·2	0·248	36·6	70·5	34·4	0·0076	0·0092	0·83
III	6000	42·9	0·256	36·4	71·5	37·2	0·0075	0·0068	1·10
III	7020	44·4	0·272	37·8	92·0	49·5	0·0071	0·0073	0·93

TABLE 7. *Coefficient of Transfer of Heat from Pipe to Brine*

Weight flow of brine (lb./hr. sq. ft.)	Temp. of Brine (°F.)	Reynolds No. (R.)	Nusselt No. (Nu.)	k_h
455,000	40	6400	59·0	0·00043
366,000	41	5100	51·5	0·00047
290,000	41	4100	43·0	0·00049
196,000	42	2700	34·2	0·00059

The average value of the ratio k_h/k_w from all observations is 1·01 for Cooler I, 0·98 for Cooler II, and 0·97 for Cooler III.

The conductance of heat from the brine to the walls of the pipe, from measurements on Coolers I, II and III, is given in Table 7. From the pressure drop through a single grid without mixing pot, the value of the coefficient of surface friction, k_f is 0·0125. This value is constant for the range of Reynolds Numbers covered, namely from 6,000 to 18,000.

DISCUSSION

Transfer of Sensible Heat

The data from the present experiments, taken as a whole, agree with the data obtained by Pierson [8] and Huge [9]. In the case of Coolers I to IV, for

which pipe surface temperature was measured, the values of k_h are systematically lower than those from Grimison[2]. This is reversed for Coolers V, VI and VII, which suggests that the calculated values for surface temperature are not quite correct.

In the experimental coolers, the actual average surface temperature may be slightly less than the measured values because heat would be conducted from the exposed pipes to the bends which were insulated. There is no evidence of this from the measurements but the points of measurement were 18 in. from the wall. The probable error is small. It would have the effect of increasing the value of k_h to above its true value. The effect on k_w, so far as can be ascertained will be nearly the same, so the ratio of k_h to k_w should not be affected.

The results from Coolers I, II and III give the ratio of the coefficients of heat transfer for shallow banks of pipes to banks 10 pipes deep. This is compared in Table 8 with that obtained by Pierson[8] for coolers of nearly the same geometrical arrangement.

TABLE 8. *Values of k_h for Shallow Banks of Pipes*

Number of Rows	1	2	3	6	10
$\frac{k_h}{k_{h10}}$ from experiments	0·73	0·81	0·87	0·95	1·00
$\frac{k_h}{k_{h10}}$ from Pierson[8]	0·64	0·76	0·83	0·94	1·00

Values of k_h for the commercial Coolers VI and VII, are some 12 per cent lower than that predicted from the data of Grimison[2], as expected for the reasons previously discussed. The reduction in performance, almost certainly, is owing to short circuit of air between the cooler and the casing and through the condensate tray. The difference between the observed and predicted transfer of heat suggests that 19 per cent of the air short circuits the cooler. The resistance of Coolers VI and VII, on the average, is 18 per cent below that predicted from the data of Grimison[2]. This would suggest a short circuit of 10 per cent. A 10 per cent short circuit would cause the coefficient of heat transfer to be reduced by 6 per cent instead of the 12 per cent actually observed. The reason for the difference, probably, is the increase in resistance caused by the spacer bars which, because of poor thermal contact with the pipes, would have but little effect on the transfer of heat.

The values of the coefficient of transfer of heat between the brine and the pipe wall on the average are 12 per cent greater than the values predicted from the standard equation given by McAdams.[7] The coefficient of resistance is more than double that for a smooth pipe, and is constant over the range of Reynolds Number covered in the experiments. These results are consistent with a pipe of considerable roughness, although the bends probably contribute significantly to the increases in resistance and heat transfer.

Transfer of Latent Heat

The values of k_w, the coefficient of transfer of latent heat, show a considerable scatter. This was owing primarily, it is believed, to error in measuring vapour pressure, but also to the indirect method used to determine the rate of condensation. Alternative methods were investigated and found to be unsatisfactory. Direct measurement by collection of the condensate was found to be impossible because water was lost by being blown off the pipes, and the residue of water which could not be drained from the condensate tray was a significant fraction of the total condensate.

In some of the experiments, the rate of condensation was determined by measuring the rate of flow of air and its change in vapour pressure. The change in vapour pressure was small and the possible error in the measurement large by comparison. The results were unreliable, as shown by the considerable degree of scatter under conditions when the rate of condensation was known to have been constant.

In deriving values of k_w it has to be assumed that the temperature of the surface exposed to air is the same as that measured at the surface of the pipe. Actually the surface is covered by water in the form either of a continuous film or a large number of small droplets. The temperature of the exposed surface, therefore, may be appreciably higher than that of the pipe. The temperature at the surface of the water cannot be measured, nor in the experiments under consideration can it be determined indirectly by measuring the thickness of the film of water. The effect is that the rate of condensation is underestimated, which, coupled with an overestimate of the difference in vapour pressure, makes the values of k_w systematically lower than they should be. The effect is probably small; for example a film of water 0·002 in. thick under the average conditions of the experiment would cause an underestimate of the value of k_w by 4 per cent.

The average value of the ratio k_h/k_w from the experiments was 0·98 as compared with the value of 1·004 predicted in the first part of this Report. Allowance for the influence of any water film has the effect of increasing the difference between these two figures. In designing coolers the ratio should be taken as 1·0 as this simplifies calculations without causing appreciable error.

EFFECT OF FROST

The series of experiments in which the effect of a deposit of frost on the pipes of the cooler was investigated, involved some change both in measurement and in operation. The method used in the earlier experiments for determining the rate of condensation could not be used because the temperature of the surface of the frost was unknown. Instead, the rate of condensation was determined by measuring the rate of injection of steam required to maintain constant humidity. A check was made at the end of the experiment by weighing the total amount of condensate thawed off the cooler. The former method would tend to over-estimate the rate of condensation because of leakage of air, and the latter to underestimate the amount of condensate because all the water could not be drained from the tray. Nevertheless, agreement was satisfactory, the difference ranging from 50 lb. when the

total amount of condensate was 1,000 lb. as on Coolers VI and VII, to 3 lb. when the total was 20 lb. as in some experiments on Cooler II. The thickness of the layer of frost was deduced from the circumference of the frost-covered pipe on the assumption that the covering formed a cylinder concentric with the pipe. Measurements could be made only on pipes at the inlet and outlet ends of the cooler.

Method of Experiment

For this series of experiments, conditions were adjusted so that frost formed over the entire surface of the pipes.

As mentioned earlier, the effect of frost is to insulate the pipes and to change progressively the geometry of the cooler. Steady conditions of running would not therefore be maintained. As it was impossible to reduce the temperature of the brine in conformity with the deteriorating performance of the cooler, this and the load on the cooler were maintained constant so that the temperature of the air increased as the experiment proceeded. The increasing resistance of the cooler to air flow as the pipes frosted reduced the quantity of air in circulation. The weight flow of air per unit area through the most restricted part of the cooler, however, changed but little.

Calculation of Thermal Conductivity

The thermal conductivity of the frost has been calculated by using the data of Grimison(2) to calculate the temperature of the exposed surface of the frost, and equation (23) to calculate the temperature of the pipe from the measured temperature of the brine. This assumes that the frost is practically impervious to the flow of air so that the frosted cooler, so far as transfer to the exposed surface is concerned, behaves as a pipe cooler of the same geometry. The temperature of the exposed surface of the frost must be such as to give the rates of transfer of both sensible and latent heat actually observed. The thermal conductivity then is obtained by evaluating equation (24)

Results

The conditions obtaining in various experiments, the performance of the coolers, and the density and thermal conductivity of the frost are shown in Tables 9 and 10. The performance of the commercial coolers during prolonged test, under comparable conditions, is shown in Figures 8 and 9. The thermal conductivity of the frost for all the experiments is plotted against its density in Figure 10.

Discussion

Figure 10 shows that when the period of the experiments were neither very short nor very long, the conductivity of the frost is practically the same as that of natural snow of the same density. Presumably the deposit in the short period experiments—under 5 hours—was so thin that there was some flow of air through the deposit; this would be consistent with the abnormally high values of conductivity. When the pipes are heavily frosted after a very prolonged period of running the conductivity is abnormally low. It is probable in this condition that the density of the frost varies from the

Cooler	Trial No.	Total Steaming Time (hr.)	Time (hr.)	Air Flow (cu.ft./ min.)	Air Temperature (°F.) In	Air Temperature (°F.) Out	Brine Temperature (°F.) In	Brine Temperature (°F.) Out	B.Th.U./hr.	Overa Conduct B.Th.U sq. ft. hr (*a*)
VI	13	337·0	—	6600	40·6	36·9	34·2	36·5	19,450	7·0
			5·5	6600	32·3	28·0	23·4	27·0	34,400	8·4
			101·5	5310	29·1	25·6	22·9	24·7	17,000	5·9
			203·0	3980	28·8	24·4	21·8	23·4	14,300	4·6
			337·0	1775	31·1	23·4	19·3	21·0	14,850	2·6
VI	14	62·0	—	6600	40·6	36·9	34·2	36·5	19,450	7·0
			1·0	6500	35·5	29·3	21·8	27·7	57,000	9·3
			31·0	3800	31·4	26·5	21·0	24·1	29,750	5·6
			62·0	1580	35·8	27·8	21·1	23·7	25,000	3·3
VI	15	136·0	—	8690	34·3	37·0	40·6	37·6	23,000	8·2
			1·5	7800	30·5	26·8	22·5	26·5	27,500	8·2
			65·0	4840	31·9	26·7	22·2	25·6	28,200	6·4
			136·0	3820	33·4	27·0	22·4	25·7	27,350	5·5
VII	16	150·0	—	5600	41·2	37·4	34·3	37·0	21,000	8·4
			8·0	5890	30·8	27·1	23·4	26·4	25,700	9·5
			76·0	4760	28·7	24·7	20·9	23·7	23,500	7·8
			150·0	536	36·4	25·5	20·5	22·4	16,200	2·5
VII	17	269·0	—	6630	46·3	39·9	34·7	38·9	36,250	8·4
			8·0	6740	30·8	27·1	23·4	26·4	25,700	9·5
			72·0	5850	27·3	24·6	22·2	23·9	15,150	7·7
			187·0	2980	29·6	24·5	21·3	23·5	18,750	6·1
			269·0	760	32·8	23·2	17·3	19·0	14,700	2·3
VII	18	141·0	—	5600	41·2	37·4	34·3	37·0	20,500	8·2
			16·0	5525	31·3	26·8	22·6	26·1	27,250	8·4
			75·0	3400	32·4	26·3	22·3	25·3	23,200	6·3
			132·5	2380	33·6	26·6	22·3	25·4	23,550	5·9
VII	19	49·0	—	5600	41·2	37·4	34·3	37·0	20,500	8·2
			6·0	5380	29·2	23·8	18·9	22·5	31,000	7·8
			49·0	4300	30·7	24·6	19·9	22·9	25,200	6·0

Notes: (*a*) Conductance and condensation are p
(*b*) Velocity is at point of maximum restri

..ll ance ..J./ . °F.	Total Condensate (lb.)	Rate of Condensation (lb. hr. sq. ft.) (*a*)	Thickness of Snow (in.)	Velocity Through Cooler (ft./sec.) (*b*)	Density of Snow (lb./cu.ft.)	Resistance (in. W.G.)	Conductance of Snow (B.Th.U. ft./sq. ft. hr. °F.)	Reynolds No.
	—	—	—	11·7	—	0·387		12,900
						0·355		
5						0·536		
5						0·622		
	1000	0·0036	0·50	8·2	23·0	0·836	0·11	20,000
	—	—	—	11·7	—	0·387		12,900
				11·7		0·445		
5						0·782		
	540	0·011	0·40	4·7	15·6	0·938		7,400
	—	—	—	15·4	—	0·632		17,000
						0·566		
5						0·465		
5	500	0·0045	0·26	5·8	25·4	0·611		8,250
	—	—	—	12·25		0·515		9,600
				14·9		0·663		
						0·808		
	720	0·0071	0·57	3·5	18·5	1·1	0·13	5,100
5	—	—	—	14·5		0·663		11,400
				14·9		0·599		
						0·732		
						0·941		
.	887	0·0048	0·56	5·7	20·2	1·206	0·07	8,200
5	—	—	—	12·25		0·515		9,600
						0·508		
						0·532		
	540	0·0056	0·28	6·8	28·0	0·611		7,150
5	—	—	—	12·25		0·515		9,600
5						0·664		
	311	0·0093	0·19	10·6	24·0	0·946	0·1	10,000

..r sq. ft. of exposed pipe surface.
..tion between pipes assuming no short circuit.

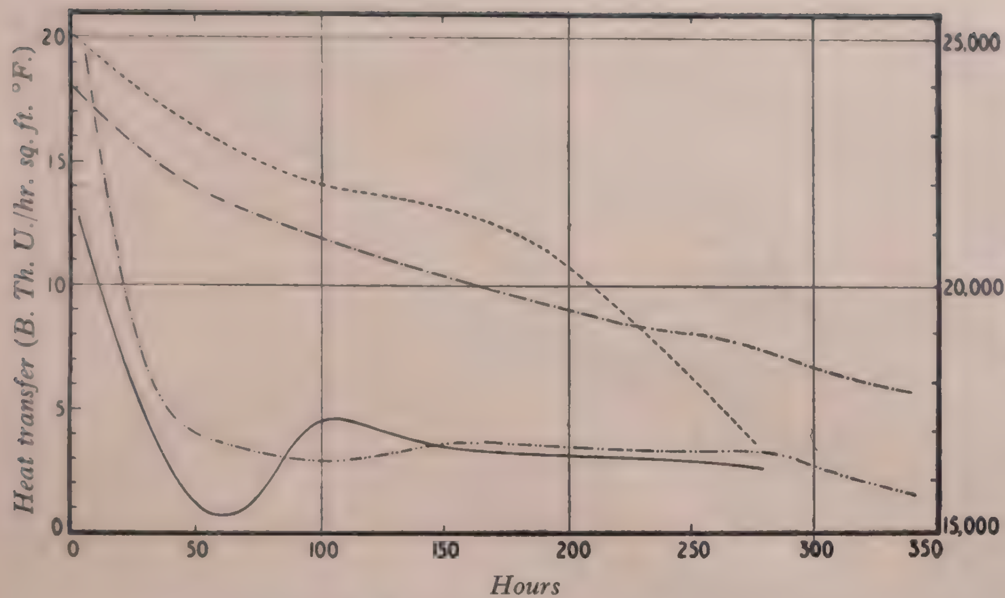

FIG. 8. *Effect of frost on commercial coolers (heat transfer)*

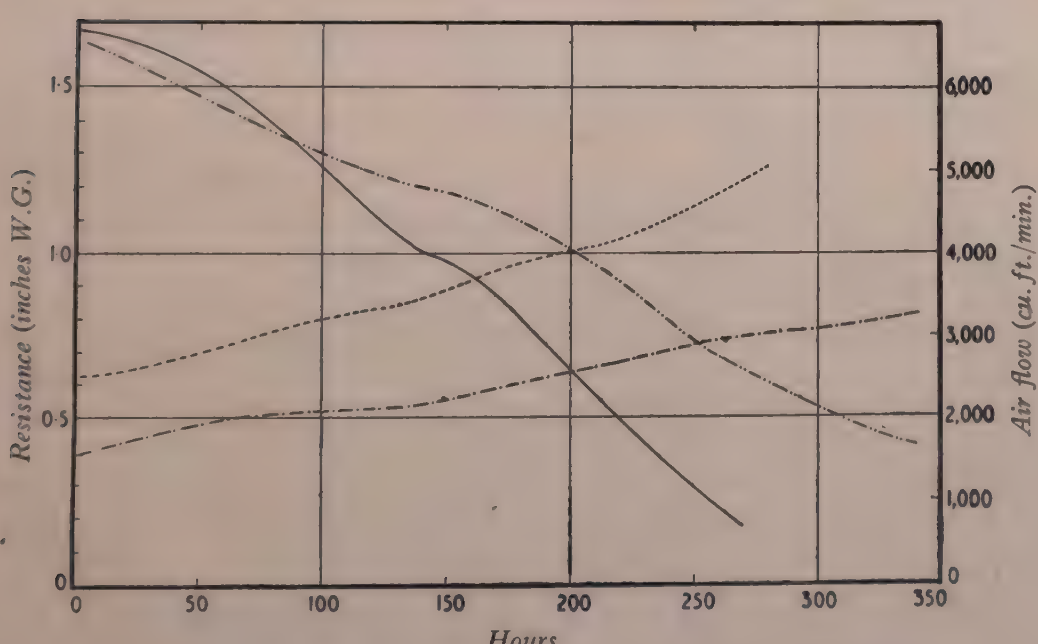

FIG. 9. *Effect of frost on commercial coolers (air flow and resistance)*

surface inwards so that the effective density, which determines conductivity, is less than the average density. The values given are average density as determined from the total weight of frost at the conclusion of each experiment.

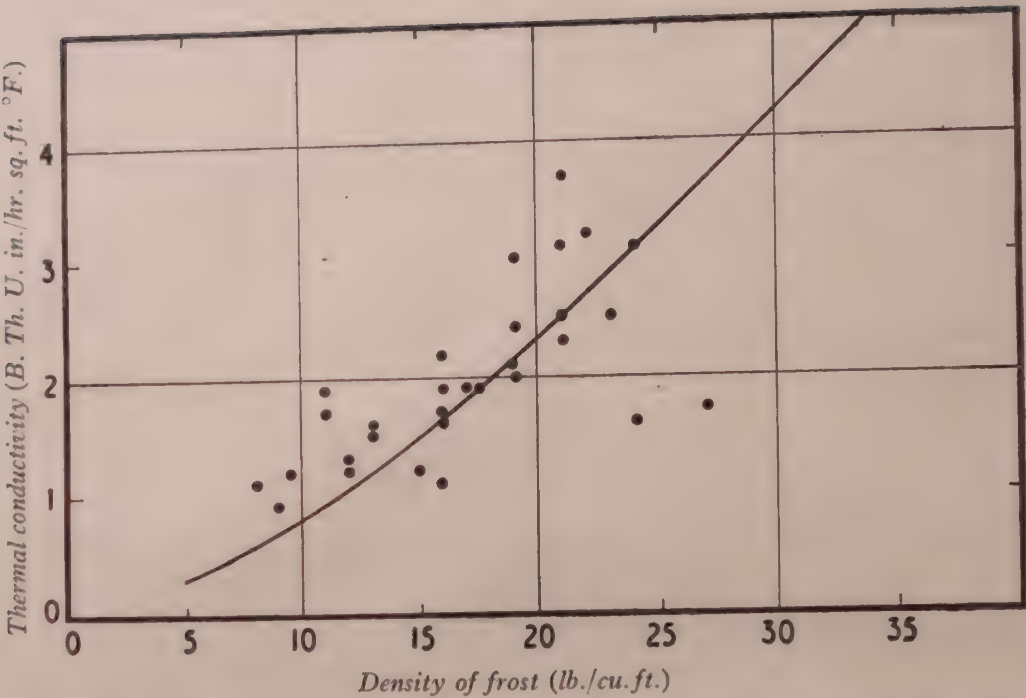

FIG. 10. •• *Experimental results* —— *Devaux's values for natural snow*

The density of the frost, it has been found, depends both on the rate of deposition and on the rate of flow of air past the pipes. It is given, closely, by the equation

$$\rho_F = C \frac{\sqrt[2]{V}}{\sqrt[3]{W}} \qquad \ldots \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad \ldots \qquad (29)$$

The value of the constant C is 1·1 for the in-line banks and 1·3 for the staggered, with ρ_F in lb./cu. ft., V in ft./sec. and W in lb./sq. ft. hr. It seems probable that this equation in conjunction with Devaux's curve, given in Figure 10, will give a reliable value of the thermal conductivity of frost over the normal period of running. The data from the present experiments are insufficient to show at what thickness of frost the calculated values for conductivity become unreliable. Further and more detailed experiments are necessary to elucidate this, as well as to find why conductivity varies with geometrical arrangement of the cooler as shown by the differing values for the in-line and staggered arrangements. Experiments with the frosting of coolers, while yielding information of interest, are not a satisfactory method of measuring the conductivity of the frost. For this, measurements on a single pipe, or other surface, are necessary and an entirely different technique must be used.

It is difficult to compare the frosting characteristics of the two commercial coolers because the limit to which frosting can be allowed to proceed is arbitrary. Clearly, the ultimate endurance of the cooler with 1½ in. pipes is greater because the surface area, as compared with the 1 in. pipe cooler, is 20 per cent larger and the passages between pipes are wider.

In the longest runs with two coolers under comparable conditions, the rate of heat transfer has diminished to one-third its value initially in 250

hours for the 1 in. pipes, and in 290 hours for the 1½ in. pipes. However, at 200 hours the performance of each cooler has deteriorated by almost exactly the same amount, the reduction in heat transfer being just under 60 per cent. This is about the limit to which, in practice, the cooler should be allowed to frost, as the rate of deterioration shows a marked increase at about this point. It seems, therefore, that the advantages in compactness and weight of the 1 in. pipes can be obtained with practically no sacrifice in performance.

It is safe to predict that had the total surface area of the 1 in. cooler been the same as for the 1½ in., the frosting characteristics would have been superior because the temperature of the surface would be higher and the rate of frosting, therefore, less.

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APPENDIX I

THE RATE OF FLOW OF AIR THROUGH A FROSTED COOLER

The main problem in calculating the performance of a frosted cooler is to calculate the rate at which air will flow through the cooler. In the analysis which follows, equations will be derived from which the rate of flow can be calculated provided the fan power is known, this being the actual air horse power and not the power developed by the fan motor. Whilst the analysis is concerned primarily with the performance of a frosted cooler the performance with clean pipes being known, the equations can be used also to determine rate of flow with clean pipes.

The system which will be considered is a cooler connected to a store by some arrangement of ducts, air being circulated through the cooler and store by means of a fan. Subscripts $_c$ and $_b$ will be used to denote the cooler and store respectively and subscript $_a$ will be used for the paths by which air short circuits the cooler. The resistance of each of these items can be expressed in terms of coefficients of resistance,

$$\Delta p_t = k_c \frac{\rho}{g} V_c^2 + k_b \frac{\rho}{g} V_b^2 \qquad (30)$$

in which Δp_t is the total resistance of the circuit. As the resistance of the cooler must be the same as that of the paths of short circuit round it

$$k_c \frac{\rho}{g} V_c^2 = k_a \frac{\rho}{g} V_a^2 \qquad (31)$$

If Q_t is the total volume of air circulating per second

$$Q_t = Q_a + Q_c \qquad (32)$$

As the velocity through the store, and also of the air which short circuits the cooler, is not known it is convenient to substitute for V in equations (30) and (31) from the equation

$$V = \frac{Q}{A} \qquad (33)$$

So, equation (30) becomes

$$\Delta p_t = k_c \frac{\rho}{g} \left(\frac{Q_c}{A_c} \right)^2 + k_b \frac{\rho}{g} \left(\frac{Q_t}{A_b} \right)^2 \qquad (34)$$

The power required to circulate the air is given by the equation

$$\text{Power} = \Delta p_t \, Q_t \qquad (35)$$

or by substituting from equation (34)

$$\text{Power} = \frac{\rho}{g} \left[k_c \left(\frac{Q_c}{A_c} \right)^2 + k_b \left(\frac{Q_t}{A_b} \right)^2 \right] Q_t \qquad (36)$$

By eliminating Q_c from this equation, an equation is obtained from which Q_t can be calculated, the value of Q_c being found subsequently by substitution. From equations (31), (32) and (33)

$$Q_t = Q_c \left[1 + \frac{A_a}{A_c} \cdot \sqrt{\frac{k_c}{k_a}} \right] \qquad (37)$$

So, from equation (36)

$$\text{Power} = \frac{\rho}{g} \left[\frac{k_c}{A_c^2 \left(1 + \frac{A_a}{A_c} \cdot \sqrt{\frac{k_c}{k_a}} \right)^2} + \frac{k_b}{A_b^2} \right] Q_t^3 \qquad (38)$$

Values of k_o and k_a are obtained from the data of performance with pipes clean. This condition will be denoted by the addition of $_d$ to the subscript.

The equation which gives k_o, the coefficient of resistance of the store is

$$k_o = \frac{\Delta p_{od}}{\frac{\rho}{g}} \left(\frac{A_a}{Q_{td}} \right)^2 \quad \ldots \quad (39)$$

The coefficient of resistance of the paths of short circuit, k_a, from equation (31), is given by the equation

$$k_a = \frac{\Delta p_{cd}}{\frac{\rho}{g}} \left(\frac{A_a}{Q_{a1}} \right)^2 \quad \ldots \quad (40)$$

The value of k_c can be obtained from Grimison [2] so that equation (38) will give the value of Q_t. The value of Q_c is then found from equation (37). The use of this equation is demonstrated in the numerical example given in Appendix II.

APPENDIX II

AN EXAMPLE OF THE PRACTICAL APPLICATION OF THE RESULTS

As an example of the way in which the results given in this report may be applied in practice, we can take the case of a cooler similar to Cooler VI in geometric arrangement, but having 40 rows of 1½ in. internal diameter pipes, a total run of 2,000 linear ft. and a surface area of 1,000 sq. ft.

The cooler which has a free area of 10 sq. ft. is in a closed circuit with a fan and storage chamber. When the pipes of the cooler are clean the fan circulates 7,200 cu. ft./min. against a head of 1½ in. W.G.

It is assumed that the air delivered to the cooler by the fan is at a constant temperature of 14°F.; that irrespective of other conditions the quantity of sensible heat taken up by the air between leaving and returning to the cooler is 25,000 B.Th.U./hr.; that the quantity of latent heat added to the air is constant and is equivalent to 4,800 B.Th.U./hr.; and that 10 per cent of the air passing through the cooler casing short circuits the cooler.

The problem is to calculate the conditions under which the cooler will be operating 120 hours after a start with clean pipes. Calculations of conductance and pressure drop are based on the data of Grimison.[2]

Initial Conditions with no Frost on the Pipes

The velocity in the free area of the cooler is 12 ft./sec. and the pressure drop is calculated from the data of Grimison[2] to be 0·52 in. W.G.

The resistance of the rest of the circuit, which includes distributing ducts and the storage chamber, is taken as being 0·98 in. W.G. when the rate of circulation is 7,200 cu. ft./min.

The value of h, the conductance of sensible heat from the surface of the pipes from the data of Grimison[2] is

8·5 B.Th.U./sq. ft. hr. °F.

The rate of transfer of latent heat, from equation(4), p. 4, is

825 B.Th.U./sq. ft. hr. in. Hg.

So the average difference of temperature between air and pipe surface will be

$$\frac{25{,}000}{1000\times 8\cdot 5}=2\cdot 9^{\circ}\text{F}.$$

and the fall in temperature of the air as it passes through the cooler will be

$$\frac{25{,}000}{7200\times 60\times 0\cdot 084}=2\cdot 9^{\circ}\text{F}.$$

The average difference in vapour pressure between air and pipe surface will be

0·0059 in. Hg.

The fall in vapour pressure of the air passing through the cooler will be

0·00512 in. Hg.

If the rate of flow of brine through each of the 3 circuits of the cooler is 600 gal./hr. and the density of the brine is 78 lb./cu. ft. then the conductance between the brine and the pipe surface will be 200 B.Th.U./sq. ft. hr. °F. (equations (21) and (22), pp. 6 and 7).

The internal surface area of the cooler is 780 sq. ft. so the average difference between brine and pipe is 0·38°F.

The conditions of operation may be summarized:

	Inlet		*Outlet*
Air-temperature	14°F.		11·1°F.
Vapour pressure of air (in. Hg.)	0·0696		0·0646
Brine Temperature (mean)		9·2°F.	

Conditions after 120 *hour running*

Frost forms at a rate of 0·0038 lb./sq. ft. hr. so the total weight after 120 hours is 470 lb. From equation (29) its density will be 29 lb./cu. ft. and its volume 16·5 cu. ft. Assuming this to be evenly distributed through the cooler the overall diameter of the pipes with their covering of frost will be 2·27 in.

The geometrical arrangment of the pipes now is 1·89 diameters between centres across the flow and 1·2 diameters in the direction of the flow.

The problem, now, is to calculate the rate of flow of air through the cooler and from this to find the conductance of heat and rate of sublimation of frost on the pipes. The problem of finding the rate of flow is analysed in Appendix I, but before the equation can be used it is necessary to know the actual air horse power which will be produced by the fan. If the characteristics of the fan are known, values of rate of flow and resistance are calculated on the basis of an estimate of the power, the estimate being subsequently revised as necessary until estimated and calculated powers agree. If the characteristics are not known, constant power may be assumed if the fan is of the axial flow type and is efficient. For the sake of illustration, constant power will be assumed. The data for the cooler with clean pipes

give the power as 955 ft. lb./sec. Other values required before Q_t can be calculated from equation (38) of Appendix I (p. 30) are

$$\Delta p_{bd} = 0{\cdot}98 \times 5{\cdot}2 = 5{\cdot}10 \text{ lb./sq. ft.}$$

$$\Delta p_{cd} = 0{\cdot}52 \times 5{\cdot}2 = 2{\cdot}7 \text{ lb./sq. ft.}$$

$$Q_{td} = \frac{7{,}200}{60} = 120 \text{ cu. ft./sec.}$$

$$A_c = 7{\cdot}7 \text{ sq. ft.}$$

$$Q_{ad} = 12 \text{ cu. ft./sec.}$$

$$\rho = 0{\cdot}084 \text{ lb./cu. ft.}$$

The value of k_c, the coefficient of resistance of the cooler, is not the same as that of the friction factor f as used by Grimison[2], because of the difference in units. The conversion factor is 2·0 so that

$$k_c = 2\,N.f.$$

with N the number of rows of pipes in the cooler. The value of k_c from the data of Grimison[2], estimating the value of Reynolds Number to be 18,000, is 6·56.

From equation (39), $k_b = 0{\cdot}1357\, A_b^2$ and from equation (40) $k_a = 7{\cdot}19\, A_a^2$. Equation (38) can now be evaluated. The value of Q_t is found to be 117·7 cu. ft./sec. (7,060 cu. ft./min.) equation (37) and from the value of Q_c is 6,280 cu. ft./min. so that 780 cu. ft./min. short circuits the cooler.

The total resistance Δp_t is power$+Q_t = 955 + 117{\cdot}7 = 8{\cdot}11$ lb./sq. ft. (1·56 in. W.G.).

By use of the new velocity and configuration of the cooler the conductance from the frost surface to the air from the data of Grimison[2] is 8·5 B.Th.U./sq. ft. hr. °F. From Figure 10, the conductivity of the frost layer will be 4 B.Th.U. in./sq. ft. hr. °F. so the conductance of the frost layer will be 24·0 B.Th.U./hr./°F./sq. ft. of external surface.

The coefficient of transfer of heat from brine to pipe will be unchanged at 100 B.Th.U./sq. ft. hr. °F. From these figures the temperature differences will be

Air to snow surface	2·5°F.
Snow to pipe	1·3°F.
Pipe to brine	0·4°F.
Total	4·2°F.

thus the required mean new brine temperature will be

$$12{\cdot}5 - 4{\cdot}2 = 8{\cdot}3°\text{F.}$$

The transfer of latent heat from the frost surface will be at the rate of 890 B.Th.U./sq. ft. hr. in. Hg. and the vapour pressure difference between the frost surface and the air will be 0·00434 in.

The fall in temperature of the air in passing through the cooler will be 3·2°F., and the fall in vapour pressure 0·00571 in.

The conditions may be summarized:

	Inlet	Outlet
Air temperature	14·0°F.	10·8°F.
Vapour pressure of air (in. Hg.)	0·0696	0·0637
Mean brine temperature	8·3°F.	

APPENDIX III

CONVERSION TABLE

For the assistance of readers familiar only with the metric system or preferring to use this system, the factors for converting British to metric units are given below:

Temperature	°F.	$=32+\frac{9}{5}$(°C.)
Length	1 ft.	=0·3048 m.
Mass	1 lb.	=0·4536 kg.
Area	1 sq. ft.	=0·092903 sq.m.
Stress	1 Lb./sq.ft.	=4·882 Kg./sq.m.
Heat	1 B.Th.U.	=0·2519 kcal.
Thermal Conductivity	1 B.Th.U.ft./sq.ft.hr.°F.	=1·488 kcal.m./sq.m.hr.°C.
Thermal Conductance	1 B.Th.U./sq.ft.hr.°F.	=4·881 kcal./sq.m.hr.°C.
Viscosity	1 lb./ft.sec.	=14·882 g./cm.sec.
Density	1 lb./cu.ft.	=16·018 kg./cu.m.
Diffusivity of vapour	sq.ft./sec.	=0·0929 sq.m./sec.

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(*b*) MISCELLANEOUS PAPERS. These include papers which explain to the technician the implications and applications of the results of the scientific researches published in scientific journals.

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(*e*) LEAFLETS. These are clear, brief statements, for the practical man, of knowledge and technique in various aspects of food preservation, storage and transport.

(*f*) ANNUAL REPORTS OF THE FOOD INVESTIGATION BOARD. These reports give a general account of the various activities of the Food Investigation Organization during the year under review, and are divided into two parts: first, the report of the Board surveying the main developments during the year; second, the report of the Director of Food Investigation which gives, in the form of a connected review for the non-specialist, the progress of the work. Up to 1939, the Report of the Director of Food Investigation was longer and recorded in summary form the data and conclusions of the researches in progress.

(*g*) The Organization also prepares FOOD SCIENCE ABSTRACTS, a bi-monthly journal of abstracts of scientific and technical publications on the processing, storage and transport of food. This is published by H.M. Stationery Office, from whom copies may be obtained.

Lists of the publications in the various series may be obtained (free of charge) upon application to the Director of Food Investigation, 20A Regent Street, Cambridge.

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